PROGRESS REPORT

FILM TRANSPORT SYSTEM

DOCUMENT No. 177

May 13, 1960

STAT

Prepared by [redacted]

Approved by [redacted]

Contributing Authors:

STAT [redacted]

Copy ___ of ___ Pages: 42

Approved For Release 2004/12/22 : CIA-RDP89B00739R000900010001-2

TABLE OF CONTENTS

I STATEMENT OF REQUIREMENTS

The photographic detecting capabilities of the film transport system must meet the following essential requirements:

1. Film Velocity

   a. Film velocity in the transverse direction must be synchronized with the scanner image velocity to within ± 0.0045 in/sec.

   b. The allowable displacement of the film from the image in the image motion compensation (IMC) direction must be within 0.004 in.

2. The focal plane on the film must not deviate more than ± 0.0005' from the true position.

3. The photographic integrity of the film must be preserved.

To insure that the above requirements are met, a film transport system must not degrade the performance of the optical system or the stabilization system by introducing extraneous and deleterious disturbances.

II SUMMARY OF ACTIVITIES

A. FILM SKEWING

Investigation of different film transport schemes revealed that the capability to turn film through a skewed angle had to be developed. When film moves over a roller at an angle other than 90° with the roller axis of rotation, friction will cause the film to climb along the roller in a direction parallel to the axis of rotation. Methods considered to eliminate or compensate for this friction included:

a) A beaded roller

b) A sectored roller

c) Pneumatic bars

Observations and tests were conducted which indicated that pneumatic bars might be suitable. A program to establish the validity of this choice was conducted. (See Appendix A for the results of this program.)

B. EDGE SENSING

Investigations of registration problems were also conducted. Sensing of film for side registration is common to all schemes. No scheme should permit physical contact between the edge of the film and the film guiding elements. Among the methods considered were several optical, pneumatic and electrical methods.

The electrical approach in which the film is used as the dielectric in the capacitors of a capacitance bridge is the one under current development. Preliminary results are very encouraging.

C. FILM GUIDING

Side guidance of film was investigated and two possible modes were established. The first is angular change of the longitudinal axis of a pneumatic bar. This will produce a change of direction in film travel which is at an angle with the original direction of travel. Second is an assembly of two parallel elements (pneumatic bars) rotated about an axis perpendicular to the plane of the film. This will produce a change of direction in film travel which is parallel to the original direction. Further consideration is required to select one of the methods or possibly both.

D. COAXIAL SPOOLS

Displacement of film from one location to another represented a severe shift in the system center of gravity and a prohibitive disturbance to the stabilization system. Investigation of different arrangements of film spools which would produce minimum center-of-gravity shift resulted in selection of a system which would have no effective shift in the center of gravity due to film relocation. This arrangement must have the supply and take up spools mounted coaxially and coplaner, thus preserving the center of gravity of an integral cassette. A breadboard model of a coaxial cassette was tested and observations indicated that the system is acceptable when used with a suitable skew turn.

E. IMAGE MOTION COMPENSATION (IMC)

A breadboard model employing two sets of parallel pneumatic elements was tested for IMC and it was determined that velocity change introduced into the film path would be troublesome. An alternate method also employing pneumatic bars and a compensating carriage which does not affect the film velocity has been designed.

F. GENERAL DISCUSSION OF SYSTEMS

A system breadboard employing pneumatic bars in all areas except metering has been assembled. Another system breadboard combining mechanical and pneumatic principles has been designed, and fabrication is in progress.

1. Pneumatic System (Figures 1 and 2)

Film was threaded from a supply spool through a pneumatic tension control device servoed to the supply spool; over two steering pneumatic bars placed 108 inches apart; through another pneumatic tension control device servoed to the take-up spool; and finally into the take-up spool. A tension level was applied to the system through a linear potentiometer and the system transported film without incident at the rate of approximately 50 inches/sec.

Tests which will include metering and guiding rollers without IMC will continue. Final observations will be made with all elements included, i.e. tension controls, metering, IMC, and steering and skew turns.

Observations from the pneumatic bar tests plus observation of the simple pneumatic system described above indicate that with proper alignment of the mechanical rollers and metering capstans in the IMC shuttle, a pneumatic system with accurate uncoerced control of film is possible.

2. Mechanical System (Figure 3)

The proposed film transport will include:

a. three tension control sections

b. a coaxial cassette employing skewed pneumatic bars

c. four film steering arrangements with associated capacitor bridge sensors,

d. two IMC shuttles and compensating carriage arrangements employing pneumatic bars,

e. two metering devices mounted in the IMC shuttles,

f. two slit controls,

g. two capping shutters.

The film will pass through the camera as follows. Film will leave the coaxial cassette passing over a skewed pneumatic bar and will be oriented by a steering arrangement into a tension control loop. The signal generated by a transducer on the control loop will control the supply spool rotation. The film will then leave the tension control device and proceed toward the forward slit. The side position of the film is again sensed and the film is steered into the slit configuration in correct attitude. When the film leaves the forward slit configuration it proceeds to a second tension control loop which is also used to maintain the correct length of film between slits. Before the film enters the rear slit configuration it is again steered to the correct attitude. After the film leaves the rear slit configuration it is transported to a third tension control loop from which a generated signal is employed to control the take-up spool. The film is then steered to the correct entrance attitude over a skewed pneumatic bar and into the take-up spool. (See Appendix B for an analysis of film velocity control.)

While the film is passing through the slit configuration it will experience IMC. The IMC motion is sinusoidal in character with a total excursion of 1.38 inches. Angular relation between IMC motion and scanner axis of rotation must be maintained within ± 40 seconds. To facilitate assembly, inspection, test and maintenance, mirror surfaces have been incorporated into component subassemblies.

An alternate method for IMC with a much reduced total excursion has been proposed and preliminary analysis suggests concurrence with the proposal Benefits derived from the smaller excursion prompt further investigation.

The smaller motion gives the following advantages:

a. Smaller unbalance to be compensated.

b. Smaller cam.

c. Larger tolerances in IMC direction.

d. Possibility of eliminating the compensating carriage.

e. Small source of disturbance to optical elements.

f. Ease of mitigation.

III CONCLUSION

Analyses supported by tests and observations are sufficiently encouraging to prompt further activity which, it is estimated, will produce a system that adequately meets the requirements. However, institution of reasonable designs, fabrications and availability of hardware components will establish a high confidence figure as applied to the estimate of a successful film transport system.

The planned program for future activities is as follows:

a. Continue pneumatic bar tests to a conclusive design. (Appendix A)

b. Conduct a complete system test with pneumatic breadboard.

c. Continue fabrication of the back-up mechanical system. This will depend on the pneumatic system. Many features, i.e. coaxial spools, metering edge, sensing etc., are common to both systems.

d. Continue magnetic tape simulation. (Appendix B)

e. Initiate simulated tests in other areas.

f. Continue and expand control efforts of reaction forces.

g. Continue and expand weight control efforts.

h. Continue and expand vibration mitigation efforts.

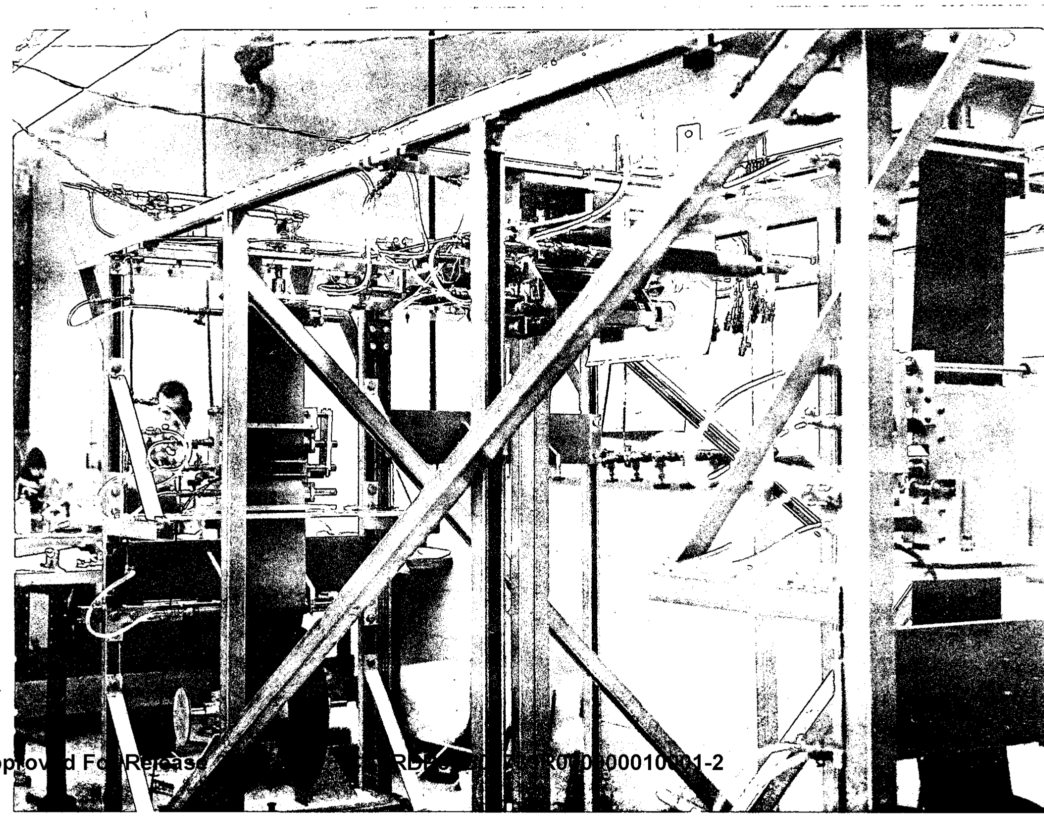
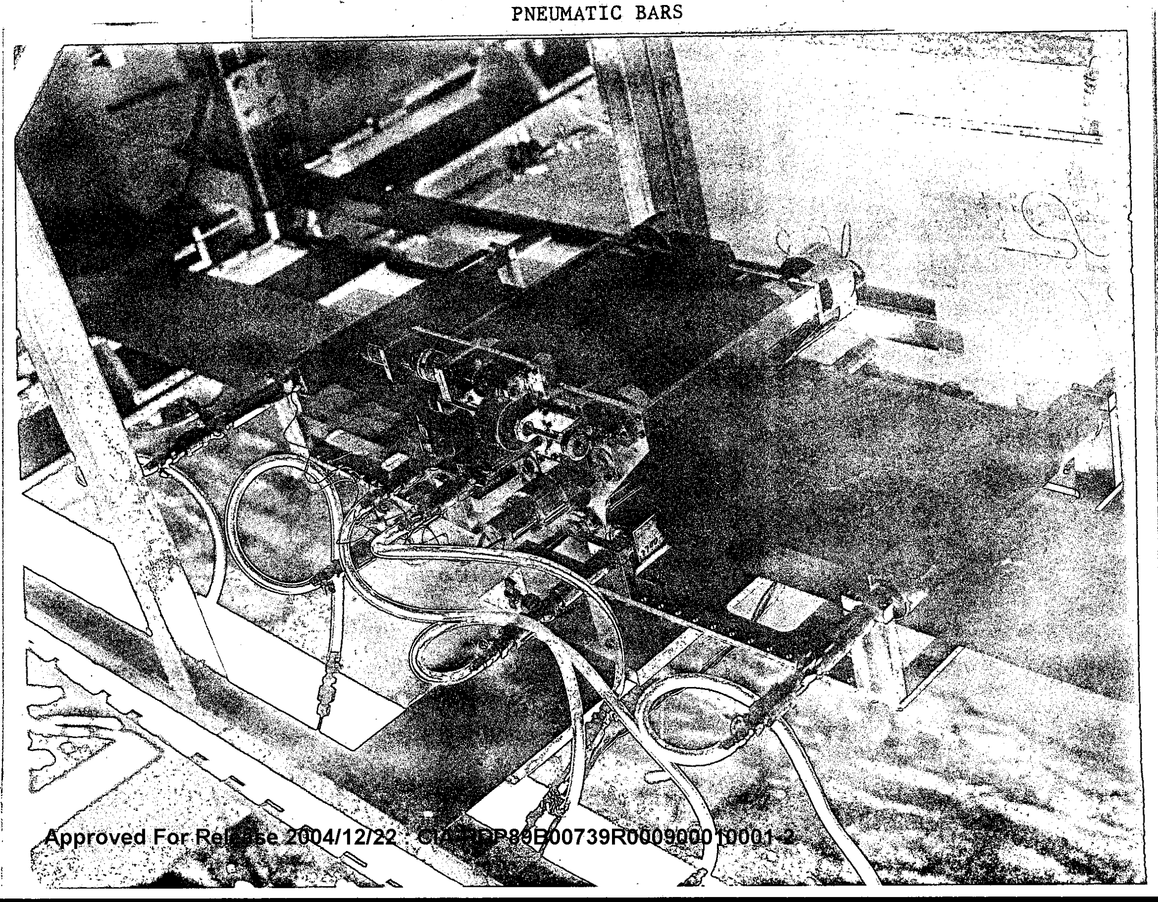
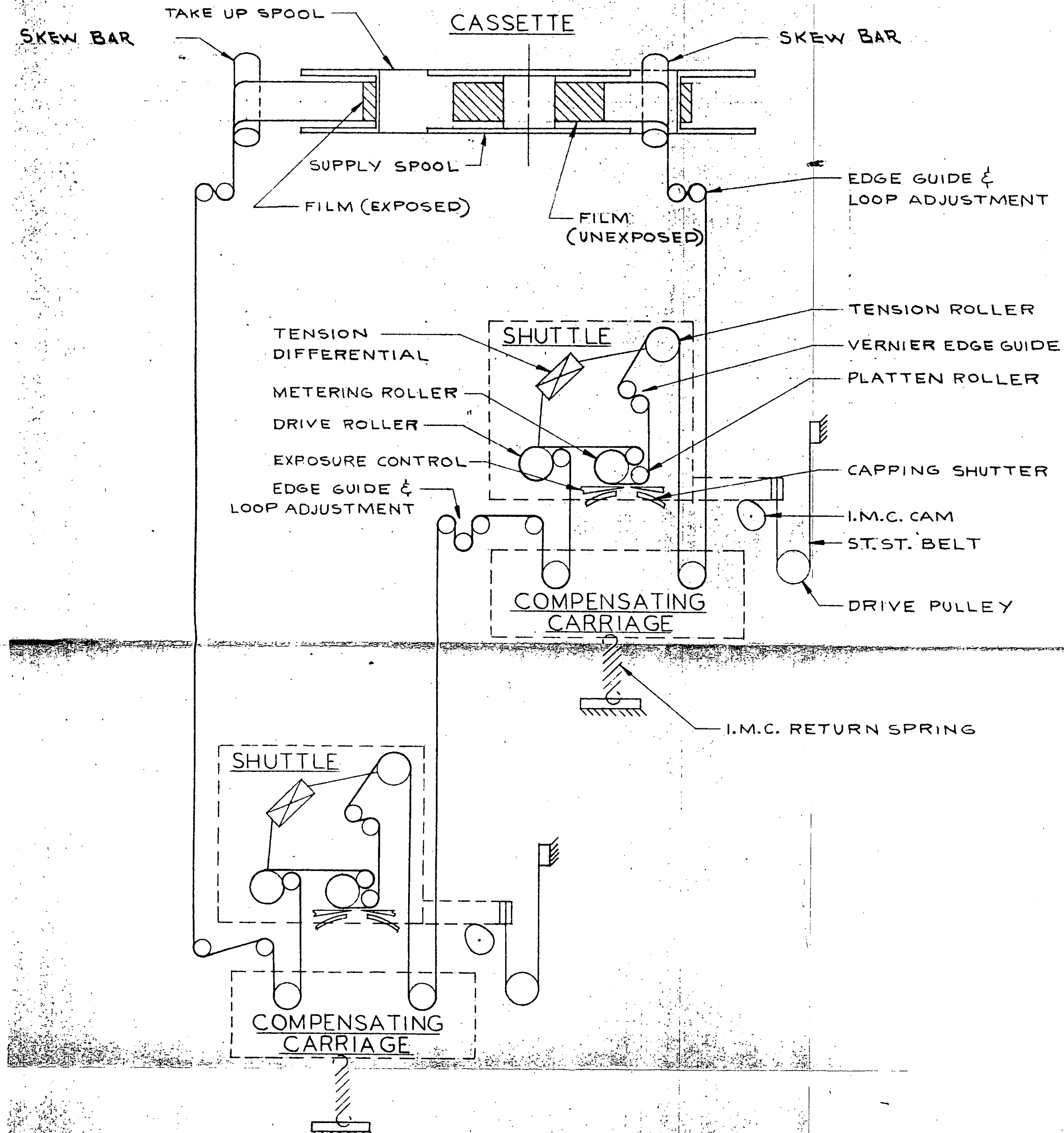


FIGURE 1. FILM TRANSPORT ASSEMBLY, PNEUMATIC BARS

FIGURE 2. IMC CARRIAGE - FILM TRANSPORT ASSEMBLY, PNEUMATIC BARS





FILM PATH LAYOUT-
MECHANICAL SYSTEM

FIGURE 3

APPENDIX A

PNEUMATIC SUPPORT IN FILM TRANSPORT SYSTEM

The characteristics of a film transport system using pneumatic guides instead of rollers, are particularly advantageous when certain of the system requirements are considered:

1. The film is protected from the abrasion damage which can result from contact with supporting and guiding members.
2. The film is free to move without being coupled to rotating rollers having inertia, friction, eccentricity and bearing problems.
3. The pneumatic guides provide effective filter elements of compliance and damping. Control of film metering for proper synchronization can be achieved more readily because of the isolating effect of the pneumatic cushions. With the use of rollers, velocity disturbances are more completely transmitted throughout the transport system, through the film.
4. The film is not constrained to a strictly circular path in making a bend; under certain conditions it will follow a skew curve without unbalancing the uniform lateral distribution of tension stress necessary for control of transport. This extra degree of freedom allows the film web to travel in three-dimensional space, and to be guided with the proper image motion compensation (IMC) component, through the focal plane.

To satisfy the last of the foregoing considerations, skewing of the film is necessary. In principle this can be accomplished by use of a compound roller, containing elements on the surface which allow an axial component of film motion during rotation. Experimental work has shown that the compound roller must be bulky; furthermore, it marks the film wherever contact is made. As a solution to these problems, and to take advantage of the other benefits, a thorough going study of the feasibility of a pneumatic film support system has been undertaken.

The method of pneumatic guidance involves the radial support of the film at all points where the film is curved. The curved film, in the area of support, is rigid and straight in the transverse direction, as it is not easily stretched. In this presentation, the curved surface is considered approximately cylindrical (all straight-line elements parallel within a turn). The pneumatic member (Figure A1) consists of a transverse bar with a convex porous surface opposed to the film at the bend, and containing a plenum chamber connected to a gas supply. The porous wall assures a distributed flow into the gap between the outer surface and the film, because of a substantial pressure drop through the material in all areas. It can be seen that the design must provide uniform lift, in opposition to the force due to tension (and in the presence of variations of tension). Equilibrium is maintained by controlling the radius of curvature around the film bend, compatible with gap pressures and forces due to tension. A pressure gradient exists within the gap because of localized exhausting of gas flow at the points of tangency to the bend.

A simplified theoretical analysis of the pneumatic bar design approach follows. Further investigations have been conducted on theoretical aspects of the problem relating to transverse deformations, or end effects. No difficulties are anticipated for application of this approach to the system design.

Two Dimensional Theory

In the discussion of the general case, it is assumed that the film has unit width. The theory starts with three basic equations:

$$(1) \quad R = T/P$$

A film under tension T with a pressure P exerted on one side will bend until the local radius of curvature R satisfies equation (1). This equation expresses the static equilibrium of forces acting on the film.

$$(2) \quad \frac{dP}{dx} = \frac{12 \mu Q}{t^3}$$

A volume flow rate Q, in a direction x, of a gas with viscosity μ flowing in a region bounded by parallel walls spaced a distance t apart will suffer a pressure gradient $\frac{dP}{dx}$ given by equation (2).

$$(3) \quad \frac{dQ}{dx} = k\,(P_S - P)$$

The flow of gas per unit area through a porous material is proportional to the pressure differences, $P_S - P$, across the wall. The constant of proportionality, k, is called the porosity of the wall. Equation (3) has been experimentally verified for the walls used.

The assumptions inherent in these equations are:

1. The fluid flow is incompressible.
2. The Reynolds number for the flow is sufficiently small to neglect inertial effects.

Both assumptions will be justified later in this report.

Consider now the system shown in Figure A1, with the film separated from the porous wall by a constant distance t. If $R \gg t$, the flow Q_x may be considered to be between parallel walls and equation (2) can be substituted into (3) to obtain:

$$(4) \quad \frac{d^2P}{dx^2} = -\frac{12\mu k}{t^3}(P_S - P).$$

The flow at $x = \pm x_0$ exists into the ambient atmosphere which is at gauge pressure zero.

$$(5) \quad P_{x_0} = 0$$

The system is symmetrical around $x = 0$ so that:

$$(6) \quad \left.\frac{dP}{dx}\right|_{x_0} = 0.$$

These two boundry conditions, (5) and (6), allow the solution of (4) to be written as:

$$(7) \quad P_x = P_S\left[1 - \frac{\cosh \omega x}{\cosh \omega x_0}\right]$$

where $\omega^2 = \dfrac{12\mu k}{t^3}$

If the film is supported only by the pressure, P_x, its shape must satisfy equation (1) which now becomes:

$$(8) \quad R_x = \frac{T}{P_S\left[1 - \frac{\cosh \omega x}{\cosh \omega x_0}\right]}$$

But if t is constant, the shape of the film also determines the required shape of the porous wall, and equations (1), (2) and (3) become self consistant, if the wall is of this specified shape.

The power, W, required to support the film is the product of the supply pressure, P_S , and the total volume flow, $Q_{x_0} + Q_{-x_0}$.

$$(9)\quad W = \left[Q_{x_0} + Q_{-x_0}\right] P_S = 2 P_S Q_{x_0}$$

Clearly, the power would be minimized if P_S were small (indeed, zero) but then the film could not be supported.

If R_0 is the radius of curvature of the film at the apex of the wall, then it is found from (8) that:

$$(10)\quad T = R_0 P_S \left[1 - \frac{1}{\cosh \omega x_0}\right]$$

since for any

$$(11)\quad T < R_0 P_S,$$

P_S must be large enough to accomodate the greatest film tension, which will occur. Assuming that ordinarily:

$$(12)\quad T = T_{max}/2 \qquad \text{so that} \qquad P_0 = P_S/2,$$

then ωx_0 satisfied the requirement.

$$(13)\quad \cosh \omega x_0 = 2$$

or

$$(13A)\quad \omega x_0 = 1.32$$

x_0 is related to R_0 through equations (1) and (8) and by the required total angle through which the film bends.

Let θ_x be the angle made by the local normal to the film at x with the center line, $\not{C}$. Thus in Figure A1, $\theta_{x_0} = \pi/2 = 90°$. By definition of the radius of curvature, R_x :

(14) $dx = R_x \frac{d}{dx} \theta_x$

(14A) $\theta_x = \int_0^x \frac{dx}{R_x}$

which with equations (8) gives:

(15) $\theta_x = \frac{2}{R_0} \int_0^x \left(1 - \frac{\cosh \omega x}{\cosh \omega x_0}\right) dx$

and in particular:

(16) $\theta_{x_0} = \frac{2 x_0}{R_0} \left[1 - \frac{\tanh \omega x_0}{\omega x_0}\right]$

which for $\theta_{x_0} = \frac{\pi}{2}$ and $\omega x_0 = 1.32$ gives

(17) $x_0 = 2.28 R_0$.

Equation (15) is also useful in determining the shape of the film in rectangular coordinates.

The remaining parameters for best operation can now be optimized with respect to the following two parameters:

(18) $\mathcal{Z} = t/R_0$

and

(19) $V_c = \frac{Q x_0}{t}$.

The first parameter is simply the film separation in dimensionless form and the second parameter is the velocity of the supporting gas as it leaves the confined area.

From equations (9), (10) and (12):

(20) $\frac{W}{T} = 4 V_c \mathcal{Z}$.

By substituting (7) into (3), integrating and substituting (13) and (17):

(21) $$T = \frac{V_c}{z^2} \cdot \frac{6\mu\,(2.28)}{(1.32)\tanh(1.32)} = 15.7\mu\,\frac{V_c}{z^2}.$$

For helium at $125^\circ F$ and $1/3$ atmosphere,

(22) $$\mu = 0.207 \text{ millipoise} = 4.1\times10^{-7}\ \text{slug/ft-sec}.$$

and for air at $70^\circ F$ and 1 atmosphere,

(23) $$\mu = 0.185 \text{ millipoise} = 3.7\times10^{-7}\ \text{slug/ft-sec}.$$

In both cases μ is independent of pressure.

For $\mu = 4\times10^{-7}$ slug/ft.-sec. equation (21) becomes:

(24) $$T = \frac{6\times10^{-6}\,V_c}{z^2}$$

when V_c is in ft/sec and T is in lbs/ft.

The Reynolds number at x_0 is:

(25) $$Re = \frac{\rho V_c t}{\mu} = \frac{\rho V_c z R_0}{\mu}$$

where ρ is the gas density.

For helium and $R_0 \leqslant 0.1$ ft.,

(26) $Re \leqslant 240\, V_c z$ and air $R_0 \leqslant 0.1$ ft.,

(27) $$Re \leqslant 680\, V_c z.$$

For laminar flow it is required that

(28) $Re < 500$ so that if

(29) $V_c z < 2$ for helium, and

(30) $V_c z < 0.7$ for air,

then the flow will be laminar.

For compressible flow it is sufficient that

$$(31)\quad V_c \ll V_s$$

where V_s is the velocity of sound propigation and

$$(32)\quad P_o \ll \text{————————}$$

the absolute ambient pressure.

Since $P_o = \frac{T}{R_o}$, this gives a minimum bound on the radius R_o. For $T = 6\,\text{lbs/ft}$ and $5\,\text{psi}$ ambient pressure:

$$(33)\quad R_o \gg \frac{6}{5(144)} = .0084\ \text{ft} = 0.1\ \text{inch}.$$

Because of the linearity associated with viscous flow, the movement of the film over the air cushion will not greatly effect the performance indicated by these equations as long as:

$$(34)\quad V_c \gg V_f,$$

where V_f is the maximum film velocity.

Parameter Optimization

To optimize the system parameters and justify the assumptions, equations (20), (24), (29), (31), and (34) may be plotted with coordinates. and z.

It is assumed that a seven-inch wide film moves $1.25\ \text{ft/sec}$ under a tension of 3.5 pounds. The gas is helium at $\frac{1}{3}$ atmosphere and $125\,^{\circ}F$.

The power required should be less than

thus

$$(35)\quad \frac{W}{T} < \frac{3.7}{3.5} = 1.06 \text{ ft/sec.},$$

which with (20) becomes

$$(36)\quad V_c \zeta < 0.26.$$

Equation (24) becomes

$$(37)\quad \frac{V_c}{\zeta^2} = 10^6.$$

In addition to this the manufacturing tolerance on the porous wall manufacture should not be greater than the distance t. Thus for $.001''$ tolerance $A_2 \zeta$ must be greater than 10^{-3} for $R_0 = 0.3''$

An examination of Figure A2 shows that reasonable parameters are:

$$(38)\quad \zeta = 4.5 \times 10^{-3}$$

and

$$(39)\quad V_c = 20 \text{ ft/sec.}$$

Under these conditions:

$$T = 6 \text{ lb/ft} = 3.5 \text{ lb/film width}$$

$$W = 1.26 \text{ ft-lb/sec} = 1.7 \text{ watts}$$

$$Re < 2.5 \text{ (viscous flow)}$$

if $R_o = 0.375''$

$t = .0017''$

$P_b = 1.33\ psi$

$P_s = 2.67\ psi$

$$2Q_{x_o} = .0033\ ft^3/sec = 0.20\ ft^3/min$$

$$k = 0.018\ ft^3/min\text{-}lb = 30\ in^3/min\text{-}in^2\text{-}psi$$

Note that equation (32) is satisfied and that the flow is very nearly incompressible.

<u>Experimental Work</u>

The parameters for an appropriate optimum system were selected according to theory, using a minimum permissable radius of curvature for the film bend $R_o = \frac{3}{8}''$. The contour for the pneumatic bar was computed, and a forming mandrel was shaped accordingly. Several pneumatic bars were fabricated using thin sintered nickel and nickel-stainless sheet materials. The materials immediately available had insufficient porosity, were extremely soft (and thus hard to handle), and had a wavy surface. These bars were improved by drilling many small holes to increase the air flow; the results were favorable to the extent that the use of sintered materials was temporarily dropped in favor of drilled sheet metal for experimental work.

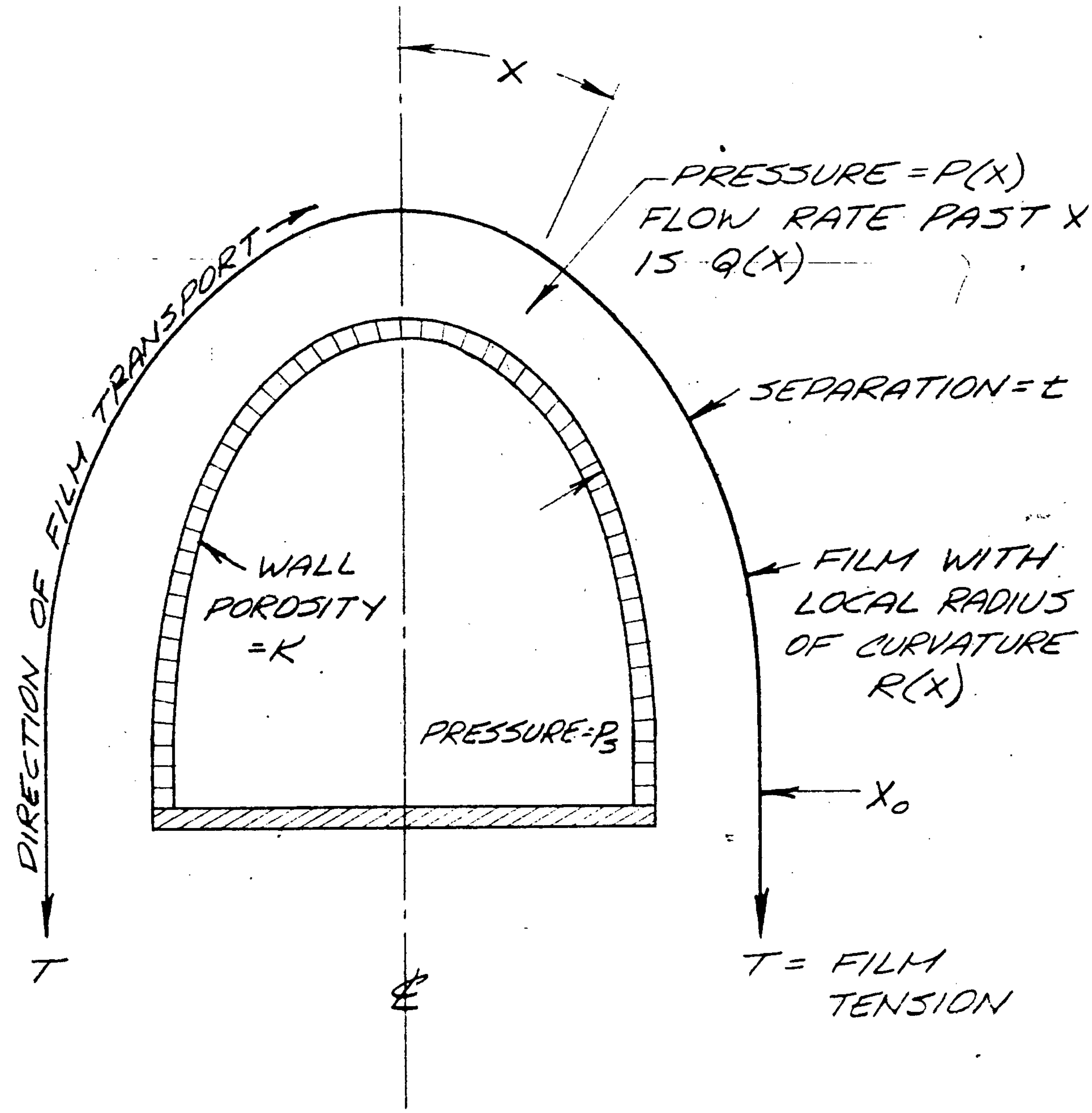
A number of sheet brass and beryllium copper pneumatic bars were formed and drilled (Figure A3), and were tested both statically and dynamically for ability to support film on an air cushion.

The static tests furnished quantitative data to determine correspondence with theory, over a range of film tensions and air pressures, (Figures A4, A5 and A6). Readings were taken of flow, and air cushion thickness over the entire area, using a microscope and transparent (fixed-out) film. (Figure A7 shows the static-film test apparatus, with an early experimental circular pneumatic bar in position.) The measurements were quite crude, owing to irregularities amounting to several thousandths of an inch in the surface contour of the handmade bars. The correspondance with theory, however, is close enough to permit very favorable conclusions as to expected performance from properly fabricated units.

The dynamic tests were undertaken to demonstrate stable and reliable performance using several of the experimental pneumatic supports with film transported continuously at normal speeds. (Figures A8 shows the dynamic test apparatus in use.) Many thousands of feet of film were passed skew-wise over the support without contact, using 180 degrees of wrap and a 90 degree lateral change of direction. The support was coated with a thin film of oil color to permit detection of contact. With reasonable care in alignment the moving film remained entirely air-supported at supply pressures as low as 3.5 pounds gage.

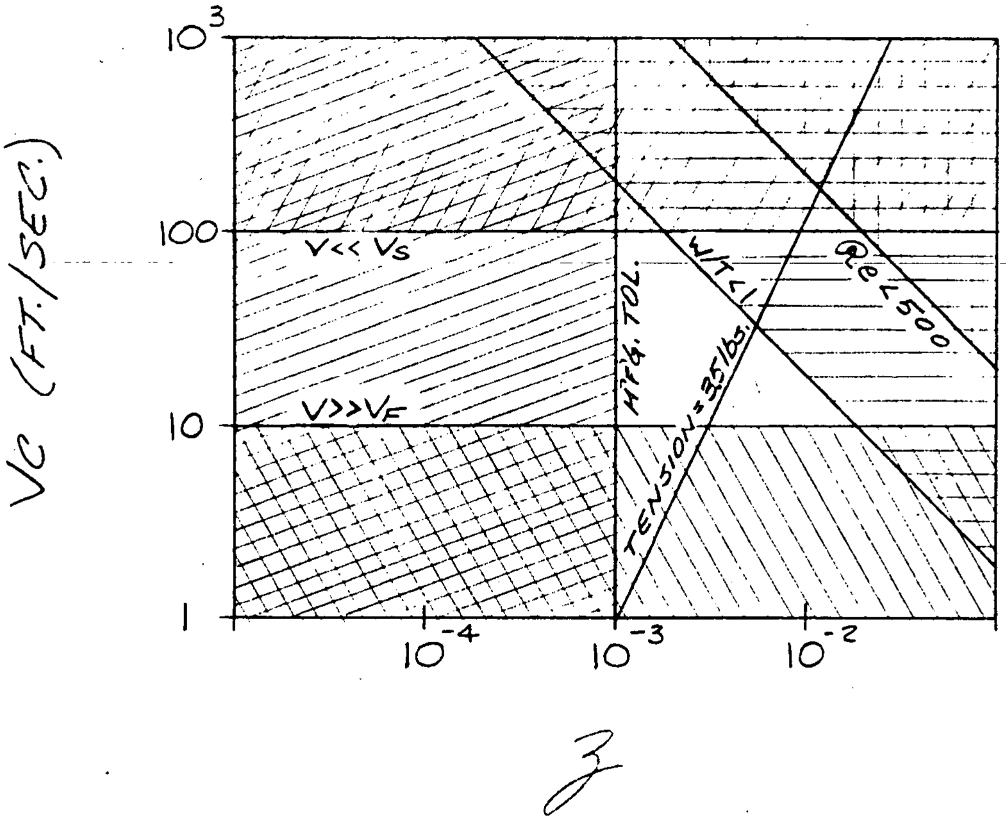
From an examination of the quantitative experimental data obtained under static film conditions, it is predicted that, with reasonable control of manufacturing tolerances for contour and straightness, the operating parameters of gap thickness, porosity, tension and pressure can be optimized for performance in close agreement with theory.

At the present time, the prototype phase of pneumatic film support development is beginning with immediate emphasis on fabrication of a number of units according to specifications. Within a short time it will be possible to predict total power requirements with accuracy. There is high probability that almost universal use can be made of pneumatic support in the transport system with reasonable ($\sim$ 100 watts) power consumption. In the event of excessive power demand, the use will be somewhat restricted, but with certain applicability for the cases of skew-bending.



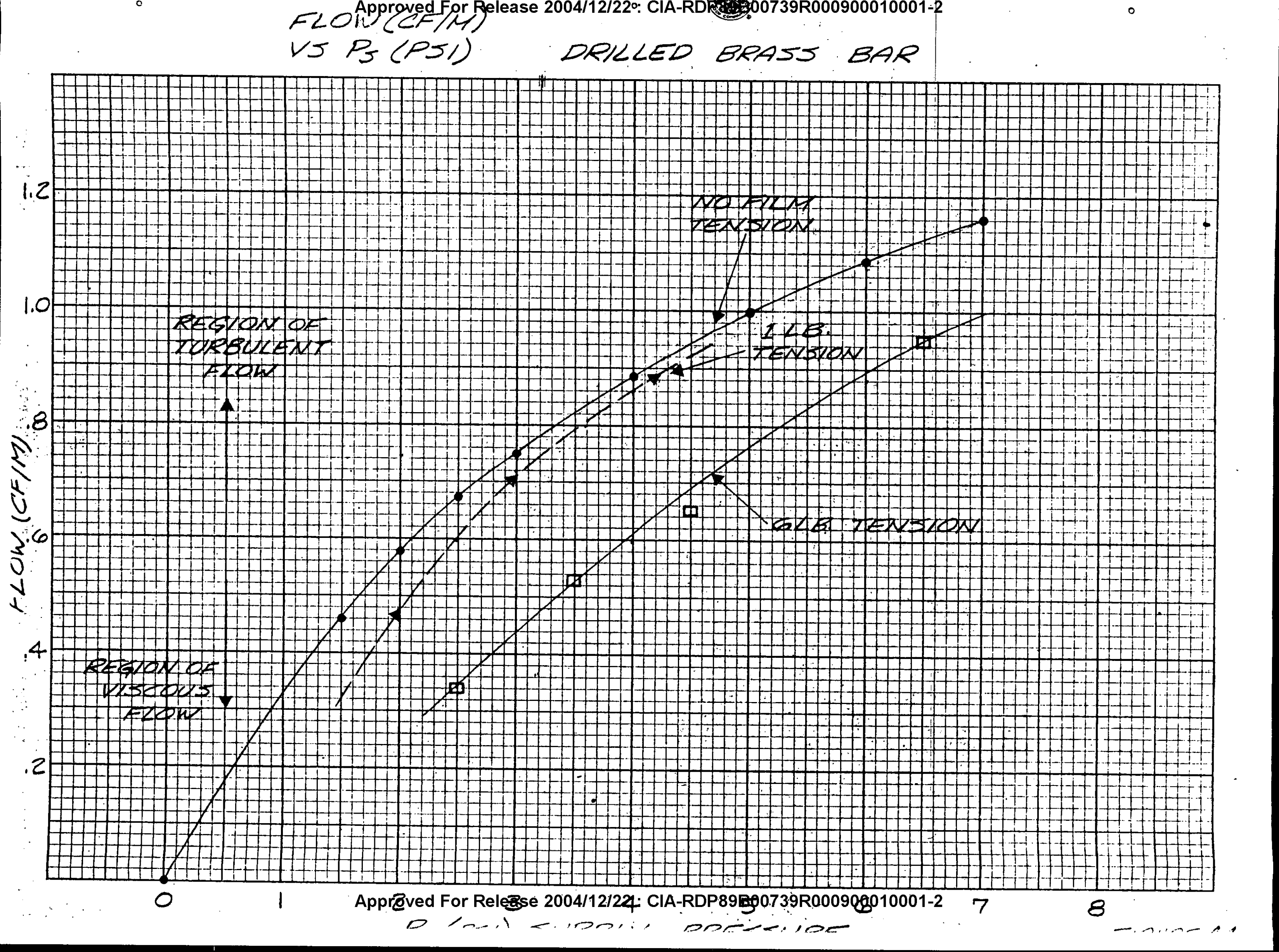
SECTIONAL VIEW OF PNEUMATIC FILM SUPPORT MEMBER

FIGURE A1



FORBIDDEN ZONES OF OPERATION

FIGURE A2


FLOW (CF/M)
VS P_S (PSI)
DRILLED BRASS BAR
NO FILM TENSION
1 LB. TENSION
6 LB. TENSION
REGION OF TURBULENT FLOW
REGION OF VISCOUS FLOW
FLOW (CF/M)
1.2
1.0
.8
.6
.4
.2
0
1
2
3
4
5
6
7
8

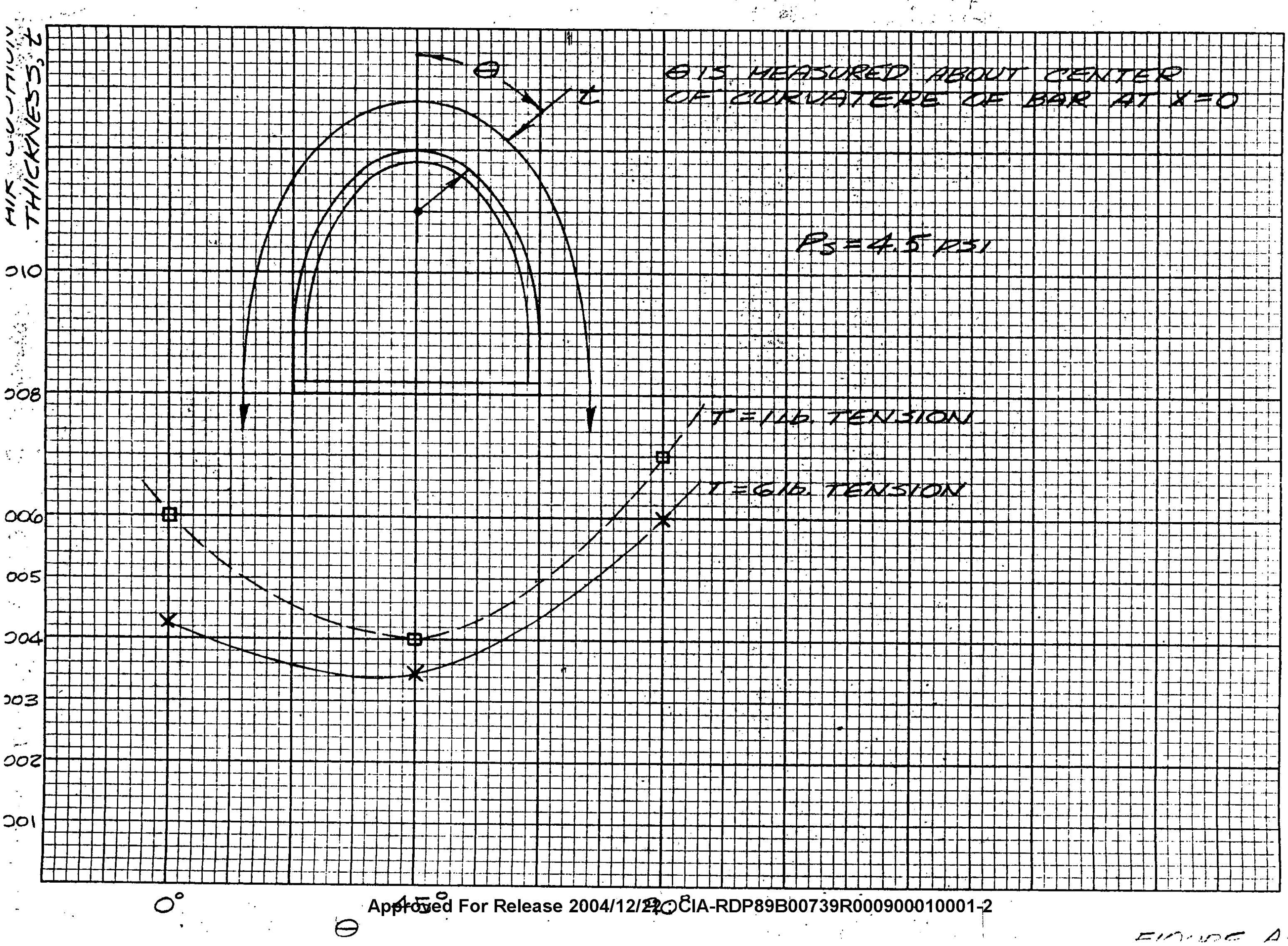

AIR CUSHION THICKNESS, t
θ
t
θ IS MEASURED ABOUT CENTER OF CURVATERE OF BAR AT X=0
P_S=4.5 PSI
T=1 lb. TENSION
T=6 lb. TENSION
.010
.008
.006
.005
.004
.003
.002
.001
0°
45°
90°
θ

FIGURE A5

FLOW VS. CUBE OF GAP THICKNESS

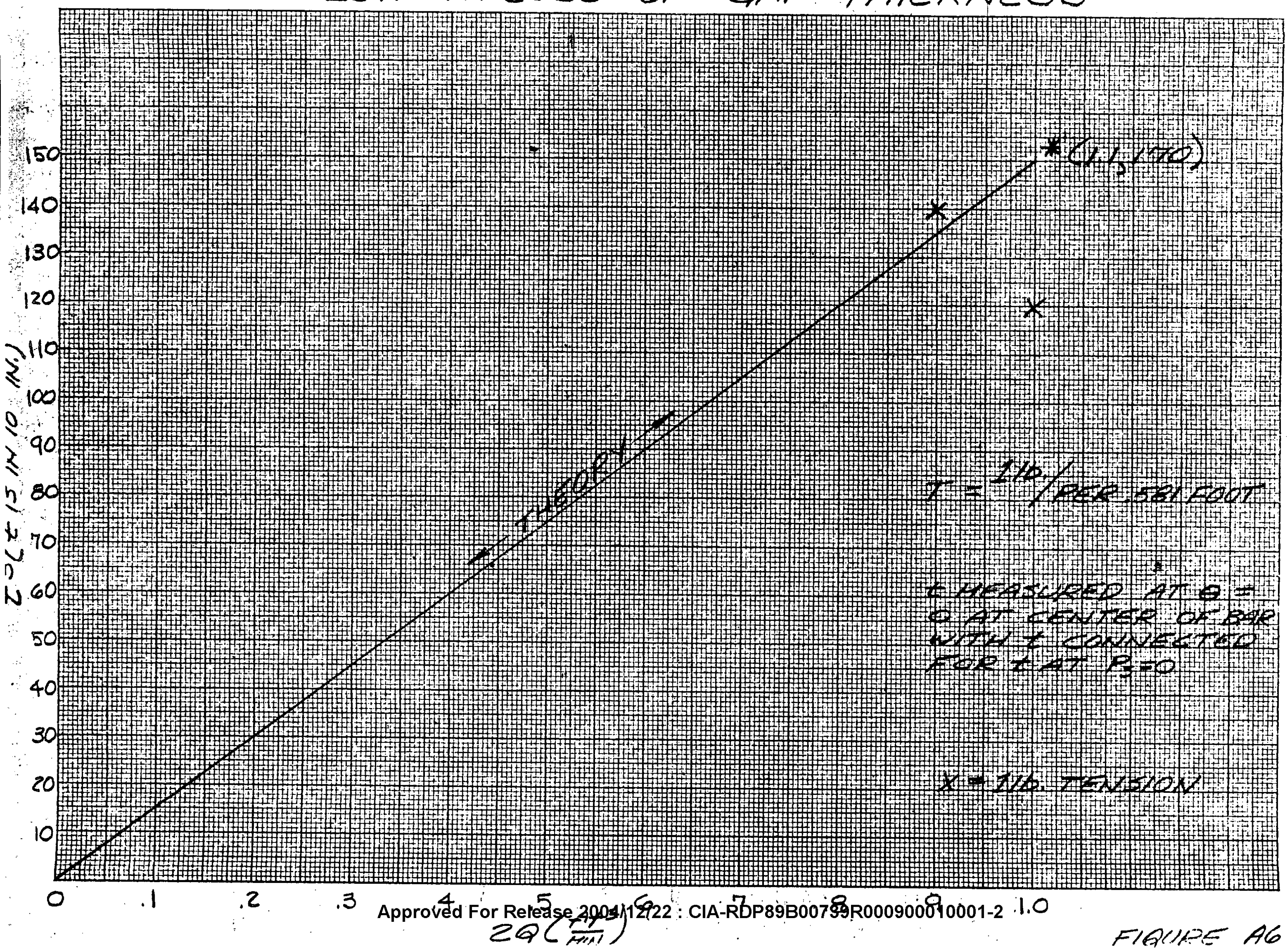


FIGURE A6

FIGURE A8. DYNAMIC TEST, PNEUMATIC BAR

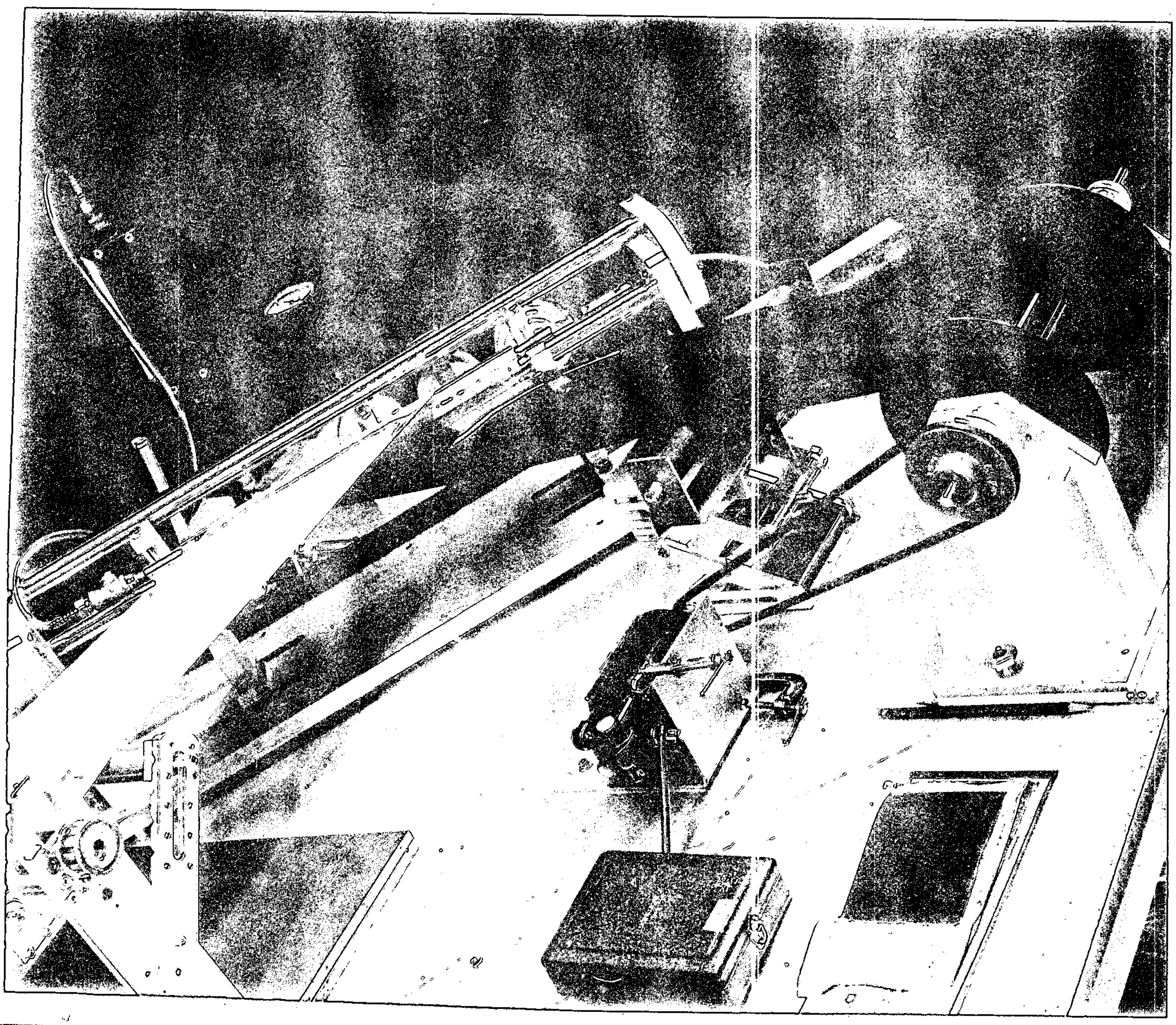


FIGURE B1. MAGNETIC TAPE TRANSPORT BREADBOARD

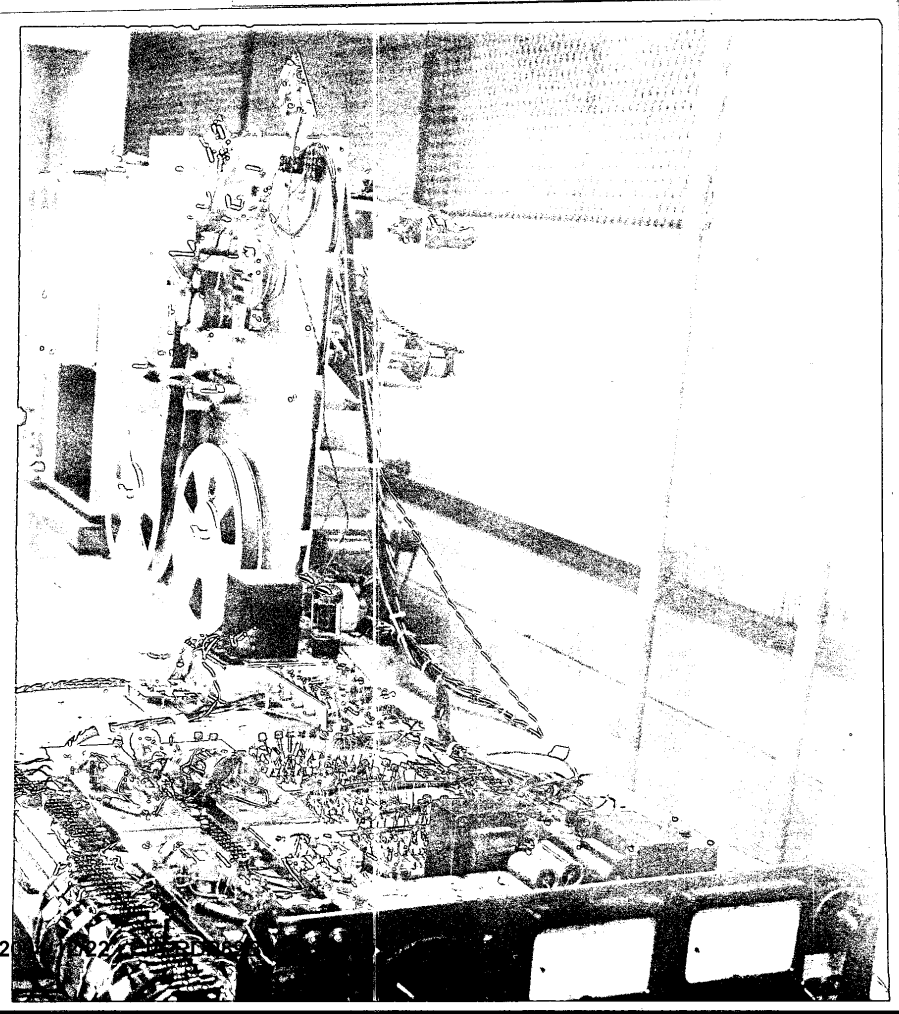


FIGURE A3. PNEUMATIC TEST BAR

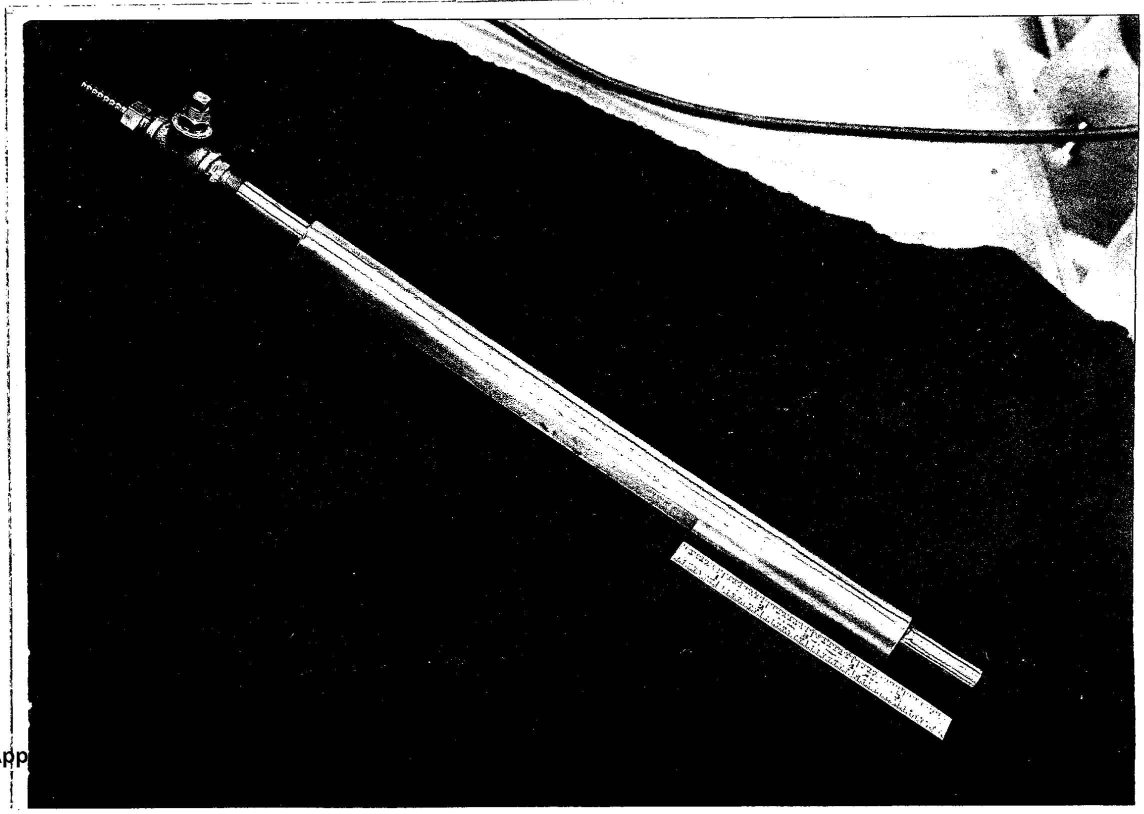
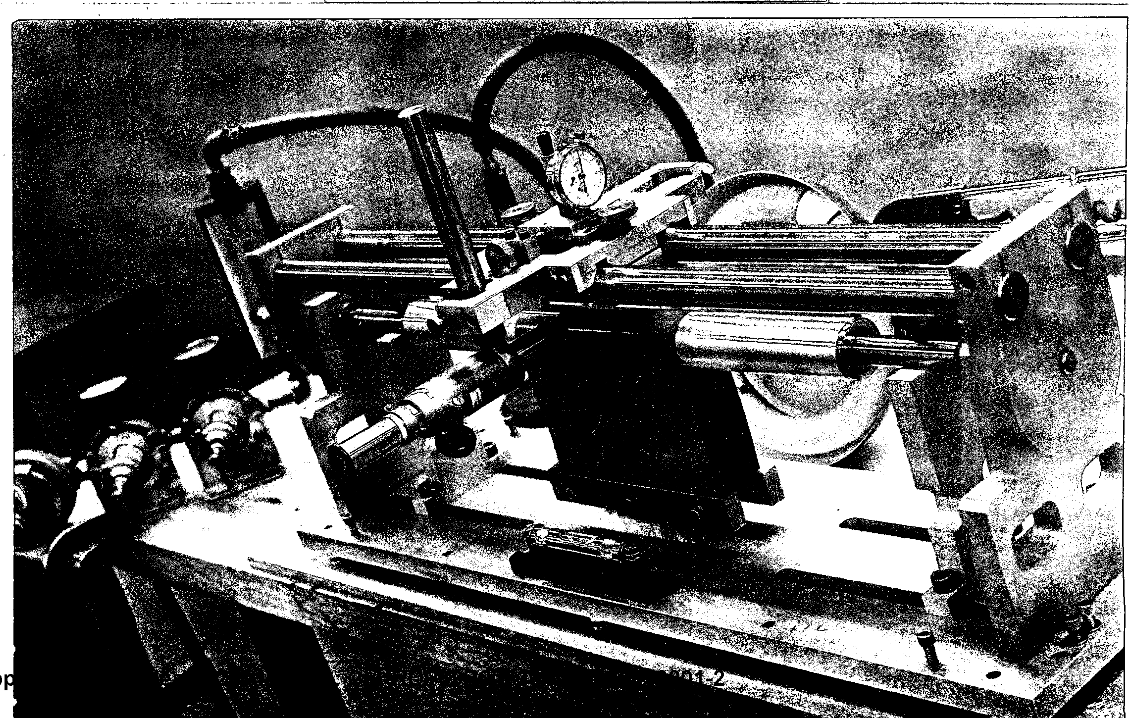


FIGURE A7. STATIC TEST, PNEUMATIC BAR



CONTROL OF FILM VELOCITY

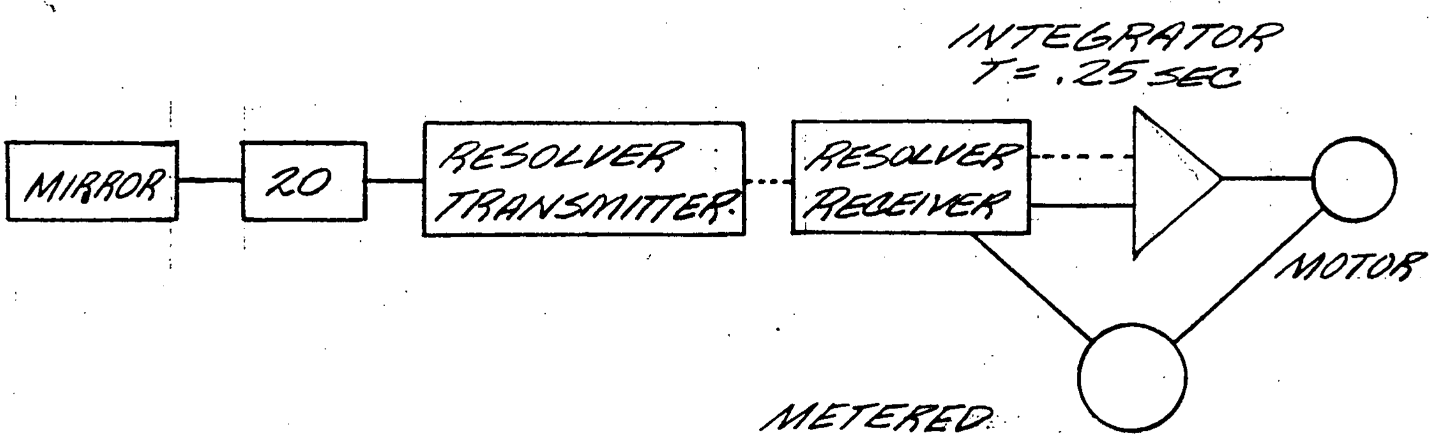
To achieve the required degree of resolution, it is necessary to synchronize the film position with the angular position of the scanning mirror. Two basic techniques of synchronization are available. One approach is to couple the scanner drive directly to the film drive, which involves the use of mechanical techniques. The most positive coupling method for this approach is a gear drive system. However the noise spectrum of a gear drive, even of the highest precision quality, is too disturbing to the overall resolution. This has been demonstrated in the development of high-accuracy, low distortion magnetic tape drive systems.

A drive system of lower noise characteristics, such as the belt or friction wheel type is also unacceptable. Such a system causes increasing power losses in the coupling and decreasing accuracy of coupling as the power transmitted increases. This results from the fact that in a direct drive system it is necessary to supply all the torque required by the film drive through the coupling member. If a belt or friction wheel drive system slips, it is difficult, if not impossible, to recover the lost displacement without reversing the direction of drive and allowing the driven unit to slip back to the correct position.

The difficulty in mechanically coupling the scanner drive and the film drive is augmented by the camera focal length. Even if a slipless, gearless drive system could be fabricated, this large distance makes it impossible to construct practical drive members without intolerable elastic deflections. These elastic deflections are manifested in inadequate tracking of the film position with respect to the scanner position. These considerations virtually rule out the possibility of using a mechanical connection between the scanner drive and the film drive.

The one alternative is to electrically servo the film position to the position of the scanner. It is therefore necessary to determine the possibility of achieving the required accuracy with the available servo equipment.

The tracking system is shown in block diagram form below:



(REFERENCE TO FIGURE B-4)

A resolver system produces an output which is representative of the angular misalignment of the scanner position and the film drive roller. At any instant there is a 1:1 relationship between the scanner position and the position of the image in the film plane. An evaluation was made to determine the maximum allowable deviation of the film from this constant image position as the film and image pass across the exposure slit.

The film travels across the slit at the velocity of 10.26 to 16.99 inches/second. The image velocity and the film velocity must be synchronized within 0.0045 inch/second. For an exposure time of 1/110 second, it is required that the displacement of the film from the image must be kept within:

$$(\pm\ 1/110)(4.5 \times 10^{-3}\text{ inch}) \triangleq \pm\ 41.0 \times 10^{-6}\text{ inch}$$

for the duration of the exposure.

The scanner turns through an angle of:

$$\text{(max. scan rate)(exposure time)} \triangleq (27.05°/\text{sec.})(1/110\text{ sec.})$$
$$\triangleq 0.2459\text{ degree.}$$

during the exposure interval.

The scanner is coupled to the transmitter of the resolver system through a friction drive, having a ratio of 20:1. Therefore, the synchro will turn through an angle of:

$$20(\text{scanner angle}) = 20(0.2459°) = 4.92\text{ degrees.}$$

The friction drive ratio can deviate from its mean value by a factor of 1/20000. For the synchro angle, this represents a peak deviation of $(0.00005)(4.92°) = 2.459 \times 10^{-4}$ degree. This peak deviation in turn represents an rms deviation of 1.7385×10^{-4} degree in the synchro shaft angle.

For the exposure interval, the synchro shaft turns through an angle of 4.92° or 1.366% of one revolution. The synchro system itself introduces an electrical error of 2.121 arc minutes rms. The rms error for this portion of a complete revolution is therefore:

$$(0.01366)(2.121\text{ arc minutes}) = 2.898 \times 10^{-2}\text{arc minute.}$$

The total rms error at the output shaft of the synchro is:

$$\sqrt{(2.898 \times 10^{-2})^2 + (1.738 \times 60 \times 10^{-4})^2}$$

$$= 3.07 \times 10^{-2}\text{ ARC MIN.}$$

The capstan drive roller that meters the film position has a radius of 1.00 inch and the radial deviation from the mean value is ±0.000010 inch. This represents a deviation of the circumferential position due to run-out of 2π (± 10^{-6} inch) = ± 62.8×10^{-6} inch, or an rms value of 44.41×10^{-6} inch for a complete revolution of the capstan.

For the exposure interval, the metering roller also turns through an angle of 4.92° or 1.366% of a complete revolution. The rms error for this angle is:

$$(0.01366)(44.41 \times 10^{-6} \text{ inch}) = 6.65 \times 10^{-6} \text{ inch.}$$

Since the output shaft of the synchro system is coupled directly to the capstan drive roller, the shaft error will result in a contribution to circumferential error. This contributive error is:

$$1 \text{ inch}\left(\frac{3.079 \times 10^{-2}}{60 \times 57.29}\right) \text{radians} = 8.966 \times 10^{-6} \text{ inch}$$

The total circumferential error is:

$$\sqrt{(8.966 \times 10^{-6})^2 + (6.65 \times 10^{-6})^2} = 11.16 \times 10^{-6} \text{ inch}$$

The film is driven by the circumference of the drive roller without slip with the result that errors in the circumferential position of the drive roller are equal to errors in the film position.

As shown above, the exposure interval the film position will have a deviation from the constant image position of:

11.16×10^{-6} inch rms,

while the requirement for resolution is:

± 41.0×10^{-6} inch

or

29.0×10^{-6} inch rms.

The requirement is therefore satisfactorily met with a safety factor of almost three.

A servo controlled breadboard has been constructed to verify the above analysis. (See Figures B1, B2 and B3.) A description of the breadboard and an analysis of the capability of the instrumentation follows.

REPORT ON MAGNETIC TAPE TRANSPORT ELECTRONICS

A. OBJECTIVE

The objective of this task was to develop a breadboard which would aid in the solution of servomechanism problems associated with the film transport system.

B. CHOICE OF MEDIUM

Magnetic recording tape was chosen as a medium to permit the recording and playback of electrical signals as a means for determining the exact behavior of the medium in the system.

Other factors in the choice of magnetic tape are its compactness, low cost, availability and reusability.

C. DESCRIPTION

Figure B4 is a functional block diagram of the transport system, including a mock-up of the scanning mirror and its drive mechanism.

Refering to Figure B4, a crystal-controlled 100 kc/sec. oscillator provides a stable reference frequency which is converted to a 50 c.p.s. signal by a series of synchronized multivibrators. This signal is amplified by A1 and drives the synchronous motor, M1, at exactly 3000 r.p.m.

The output from M1 is applied through speed reducers SR1 and SR2 to drive the mirror mock-up, a flywheel possessing the same inertia as that of the camera prism.

The synchro control transmitter, CX, is coupled to the mirror through gearing to establish the correct speed ratio between the mirror mock-up and the tape transport platens.

Position information is transmitted to the position servomechanism consisting of the synchro control transformer CT1, integrating amplifier A4, servo amplifier A3, servo motor-generator M2-G1, speed reducer SR3, and vernier drive platen, PL2.

The integrator, A4, is utilized to eliminate the short-term repetitive errors which are inherent in synchros. This lengthens the response time of the servomechanism considerably, but is not detremental to the system performance in view of the fact that there are no rapid changes in scan mirror velocity.

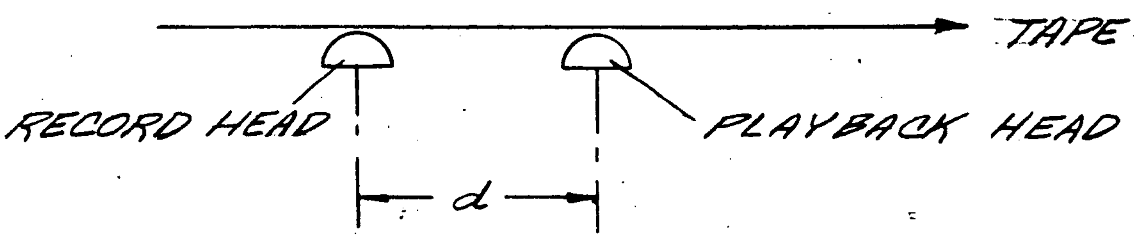
Generator G1 is used for two functions. First, its signal is summed with the integrated error signal from A4 to velocity damp M2. Second, its signal commands servo motor M5 through amplifier A6. With proper adjustment of the gain of A6, motor M5 can be slaved to motor M2 in a manner such that M5 will assume the major portion of the tape transport load. This permits the position servo to drive a relatively light load which requires small positional adjustments.

Platen PL1 is coupled to PL3 through the mechanical differential D1. A constant torque is applied to the differential spider by motor M4 through speed reducer SR5. The differential torques thus applied to PL1 and PL3 maintain a constant tape tension between the two platens without affecting the drive applied to the tape by the platens. Tension is adjusted manually by means of the variable transformer P1 which controls the power applied to M4.

A tape tension of 3 ounces at the supply reel is regulated by means of a movable tension roller which is positioned against the pull of a 6-ounce constant tension spring. The servomechanism comprised of synchro control transformer CT2, servo amplifier A5, motor-generator M3-G2, gear train SR4, the supply reel and the tape, maintains the tension roller in position. The control transformer is coupled to the tension roller and is connected to act as a position sensor for the roller.

The servomechanism at the tape-up reel is identical to that of the supply reel.

D. RECORD/PLAYBACK SYSTEM CALCULATION



d = Distance between center lines of record and play-back heads, expressed in inches.

λ = Length of one cycle of recorded material, expressed in inches.

f = Frequency of recorded signal in cps.

V_t = Tape velocity in inches/sec.

$\lambda = V_t/f$

$V_t = f \lambda$

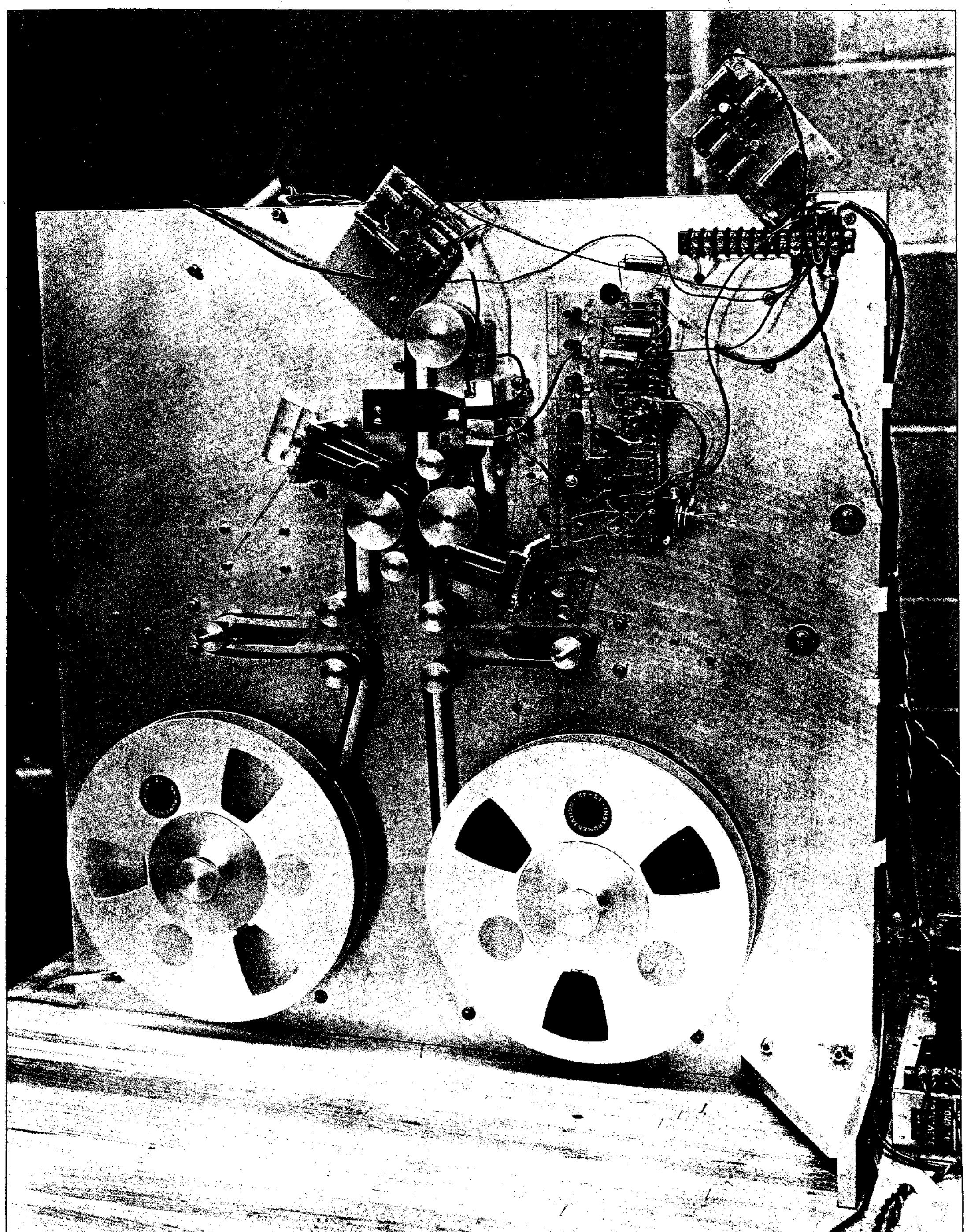
With 50 cps applied to synchronus motor M1, and a 15:1 speed reduction between the motor and CX, the CX will turn at 50/15 = 3 1/3 rev/sec. The platen will rotate at an identical speed. With a 2" diameter platen, tape speed will be $50/15(2\pi) = 20.9$ in/sec. At $f = 5,000$ cps, $\lambda = 20.9/5000 = .00418$ in. Frequency, f, will be precisely constant at 5,000 cps, but V_t will vary due to errors in the system, and the amount of variation can be determined by measuring the variation of λ. Let d = 1.3 in. which is equivalent to 1.3/.00418 = 310 wavelengths. Then, if d is adjusted so that the playback signal is exactly opposite in phase to the recorded signal, the two signals may be observed on the oscilloscope as a straight diagonal line.

A change in phase of $\pm 3°$ will be easy to detect with this type of scope pattern.

A phase change of 3° would represent a cumulative variation in λ over a distance of 310 wavelengths of $3°/360° \cong .0083$ wavelengths. The average change in λ would be .0083/310 which equals .0027%. It therefore follows that by this technique variations in V_t as small as .0027 or 27 parts per million, may be detected.

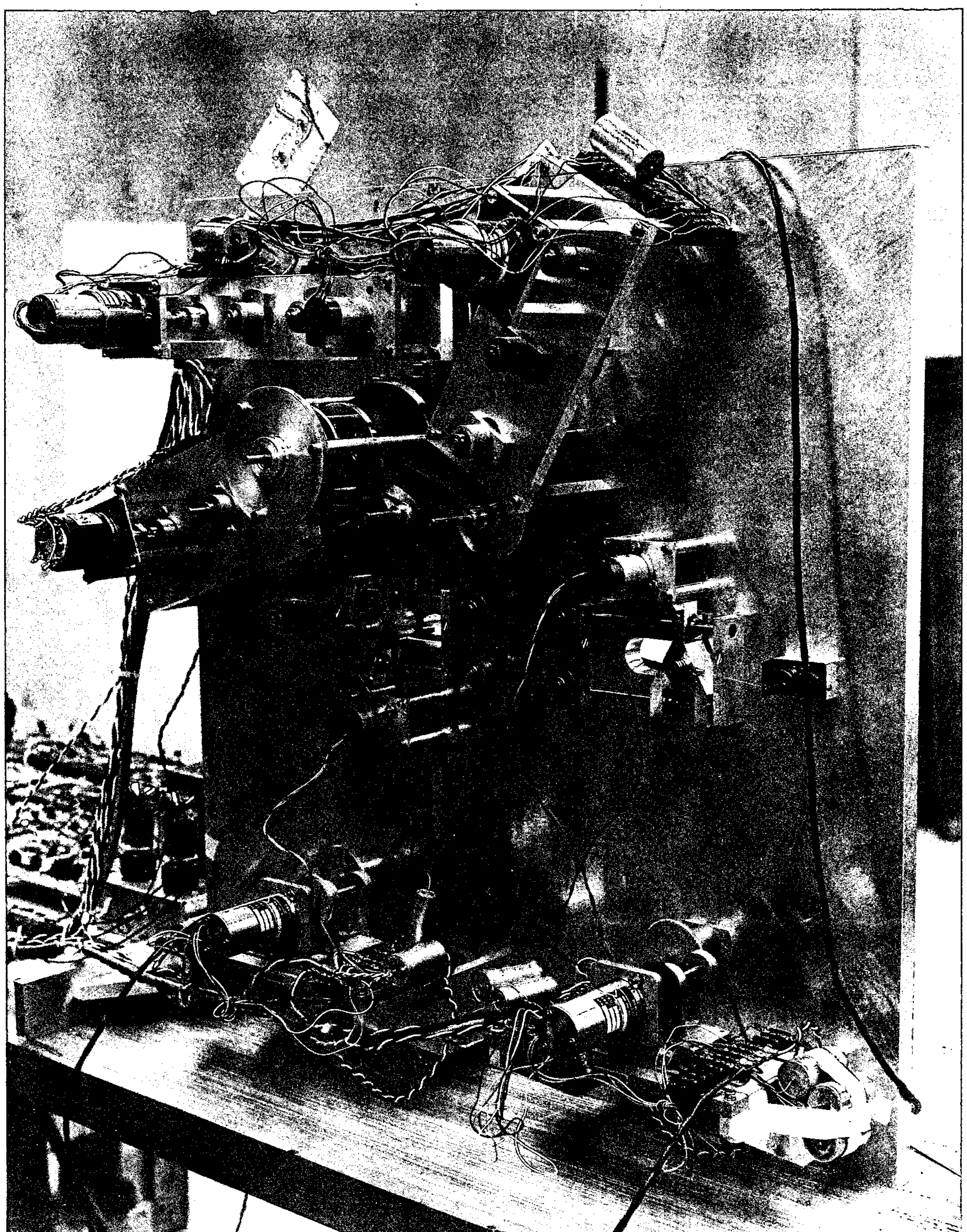
NOTE: If the spacing between the record and playback heads is equal to a wavelength of the variation in tape velocity nothing could be detected. This deficiency is avoided by testing at several recorded frequencies which are not related by integral multiples of 1/2.

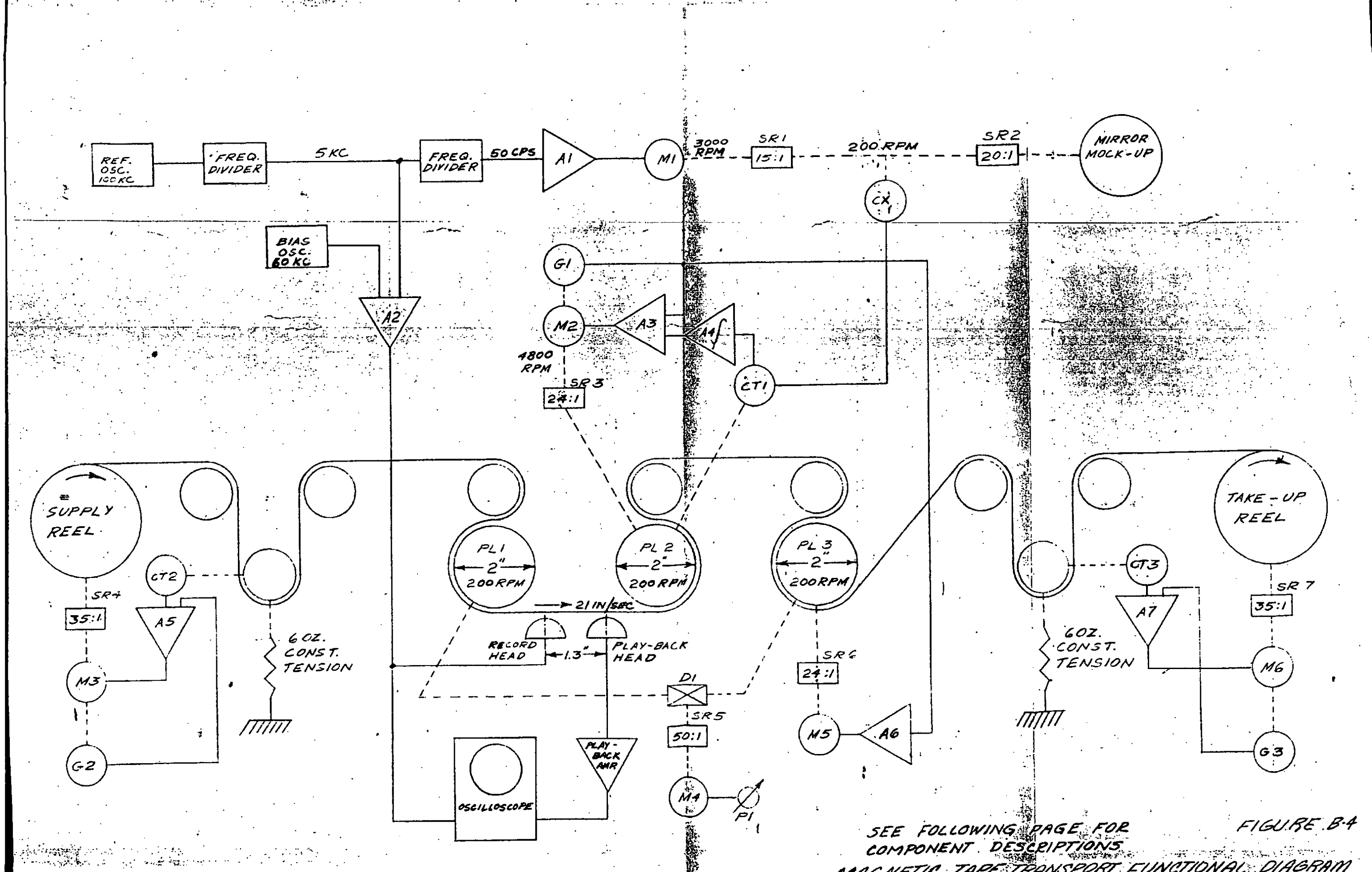
FIGURE B2. MAGNETIC TAPE TRANSPORT BREADBOARD



Approved For Release 2004/12/22 : CIA-RDP89B00739R000900010001-2

FIGURE B3. MAGNETIC TAPE TRANSPORT BREADBOARD





SEE FOLLOWING PAGE FOR COMPONENT DESCRIPTIONS

FIGURE B-4

MAGNETIC TAPE TRANSPORT FUNCTIONAL DIAGRAM

COMPONENT DESCRIPTIONS

Figure B4

Reference Designation	Description
A1	Synchronous Motor Driver Amplifier; 37-67 c.p.s.; 130 volt, 16 watt output
A2	Mixer Amplifier
A3, A5, A6, A7	Servo Amplifier and Preamplifier Mixer; 9 watt output
A4	Integrator; approximately 0.5 second time constant
M1	Synchronous Motor; 2220-4220 r.p.m.
M2-G1	Size 18 Motor Generator
M3-G2, M6-G3	Size 15 Motor Generator
M4-M5	Size 18 Motor
CX	Synchro Control Transmitter
CT1, CT2, CT3	Synchro Control Transformer
SR1	Speed Reducer, Frictional
SR2	Speed Reducer, Precision, 3 Gears
SR3, SR5, SR6	Speed Reducer, Belt Drive
SR4, SR7	Speed Reducer, Gears
D1	Friction Differential
PL1, PL3	Main Drive Platens
PL2	Vernier Platen
P1	Variable Voltage Transformer